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UC SAE Baja Steering Rack Design and Analysis

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Abstract:

The steering system within a vehicle is a vital aspect of operation and performance. The Baja society of automotive engineering (SAE) challenges students to design and manufacture an off-road vehicle to compete in a series of racing challenges. The team of engineers works together to discover challenges in design, manufacturing, and business. The steering system for the Union College Baja Team 2023 vehicle is designed as a rack and pinion mechanism, fulfilling the team designated requirement of 1.5 inches of travel per side. The mechanism has a link-to-link distance of 17 inches, and a steering wheel rotation of 172 degrees to account for the length of travel. The system consists of a round gear rack, a 16 teeth metal gear, high load ball bearing, ultra-low friction oil embedded sleeve bearings, a press-fit drill bushing, and inline ball joint linkages. One of the focus points of the steering system was the durability of the middle housing, where failure has occurred in previous years. Using a static load of 3g (2100 lbf) the housing portion of the steering system achieved a factor of safety of 7.2 discovered using finite element analysis, while the key receiving a shear stress applied by the torque of the wheel achieved a factor of safety of 1.5. The manufacturing techniques for this system include CNC machining, water jet techniques, and various assembly and fitting methods. A bearing and support system was put in place to emphasize the stability of the system. On May 4-7, 2023, the vehicle competed at the annual competition in Oshkosh, Wisconsin, with a final overall score of 208.41.

Problem Definition and Design Requirements

The Baja Society of Automotive Engineers (SAE) challenges students to design and manufacture an off-road vehicle to compete in a series of racing challenges. Each team of engineers works together to overcome challenges in design, manufacturing, and business. There are many different aspects that contribute to the success of a racing vehicle from the drive train to hubs, brakes, suspension, cabin, and steering. The success of each of these individual components many times rely on each other. For a successfully operating vehicle, each sub-system of the car must work in tandem with the others. The steering system within a racing vehicle is a vital aspect of operation and performance. During the Baja competition, the vehicle will be required to make sharp turns, while maintaining a high speed. The success of the steering system can be seen directly with performance of the vehicle itself. The car must steer, effectively and efficiently while withstanding impact and static loads from obstacles and potential collisions. Without the steering system, the vehicle cannot function properly.

There are four dynamic events within the Baja SAE competition in Oshkosh, Wisconsin, which measure the efficiency and reliability of the vehicle: acceleration, maneuverability, traction, and suspension. Within these dynamic events, each vehicle is scored on the categories of static, cost even, design, sales presentation, acceleration, maneuverability, sled pull, suspension, and endurance. The endurance test is the most important aspect of the vehicle's performance, where each team races on an intense and rough terrain track for four total hours, or until the vehicle is unable to continue to perform.

The purpose of the following research design is to model and manufacture an effective and reliable steering system for the Union College SAE Baja vehicle. In previous years, the steering system for the vehicle has been highlighted as an area of weakness and continuously varied from

year to year. The goal of this research is to create a system which is not only effective for the 2023 competition but acts as a lasting model for future Union College teams and vehicles.

There are many important design requirements which must be considered when designing the steering system. Many of which are vital to the success of the vehicle, while others are design objectives which maximize efficiency and are not vital. When considering the Baja SAE rules, the following must be kept in mind when designing the steering system. "B.4.3.3 – Positioning: … The driver must be able to reach the cockpit kill switch and steering wheel but not allow their arms to exit the cockpit. … B.7.1.4 – Brake Lines: … All brake lines shall be routed and oriented such that they are not pinched by steering or suspension parts. … B.8.7.1 – Linkages: All steering or suspension links exposed in the cockpit shall be shielded with a sturdy, robust, metal cover."[1] With these rules in mind, it is important to keep the system small enough to be able to fit within the shielding, as well as ensure a smooth transition between the system and the steering column for easy access to the wheel which will be given the input torque from the driver.

The design choices and team requirements outside of the rules and regulations of the Baja SAE rule manual are based on the specific model of the Union College vehicle design. The rack and pinion method was used as the basis of the design. In a rack and pinion mechanism, the input torque is given by the driver on the steering wheel, which is connecting to a steering column. This torque on the steering column connects to a shaft through a U-joint. This shaft in turn rotates a gear with the use of a key. The gear receives the torque, sliding the rack to the left or right. This rack is connected to the wheels and suspension system, turning the wheels to a desired angle for the appropriate turning radius. The requirements for this system include a travel distance of 1-1.5 in. each side to result in the desired 22° wheel angle, a link-to-link mounting distance of 17 in., a steering rotation of 180° clockwise and counterclockwise, and the ability to withstand shaft torque

and static loads from collision. The 22° represents the ideal turning radius for vehicle to operate at a high speed, and the suspension system can achieve this angle with only 1.5 in. of movement per side. The steering rotation of 180° to achieve this 1.5 in. of movement is based on the recommendation from the chief of drive train, who will be operating the vehicle. In recent years, failure was seen with the steering system being unable to withstand shaft loads and static torque. The steering system from last year was not able to withstand the torque from the driver, nor the load which would simulate a collision for the car. Along with the design requirements are design objectives. These objectives are to help maximize the efficiency of the vehicle. The design objectives are to be lightweight, low volume, and low cost. For a racing vehicle, it is important to not have weight where unnecessary, to maximize the speed of the car while remaining under a set budget.

Introduction

The history of steering systems continues to evolve since the beginning of the automobile industry. These systems began with relatively with the same design in initial vehicles, and moved to more complex systems, as needs and specification changed for different types of automobiles. The initial design for a steering system in the 1800's came from the inspiration of boats, as they were the only vehicle with turning capability at the time [2]. A tiller was first used to steer vehicles, but this method was increasingly flawed and ineffective. The use of the steering wheel was not implements in United States until 1899, where it became an integral part of the model T vehicle [2].

While the design of manual steering wheels continued throughout the 1900's, a major development changed the future of all steering systems. Manual steering systems were effective but required increasing force and thrust to turn with speed and size increase within vehicles.

Hydraulic Power Steering (HPS) in the 1960's was extremely impactful for the automobile industry [3]. This worked by using a hydraulic piston to generate pressure on the system and transmit this pressure to road wheels from the steering wheel input. The required force for the steering wheel turn was decreased significantly, reducing the stress and strength required for everyday drivers. The implementation of these Advanced Driver Assistance systems (ADAS) increased exponentially within the global automobile industry in the late 1900s-2000s [3]. The burden reduction on the driver was extremely impactful for the future development and manufacturing of vehicles. Today, we see the assistant systems in the majority of everyday vehicles. This also leads into the production of self-driving and steering vehicles, further lessening the burden on the driver. With autonomous operating systems also comes with a need for a relationship between the driver and the autonomous system for override and safety purposes, which is an aspect of steering technology which remains far from perfect.

While everyday cars will continue to evolve within the field of electronic steering, there are different requirements and needs for racing and off-road vehicles. The Baja SAE competition requires a vehicle which is efficient, as well as extremely durable. Electronic steering systems may be effective, but in terms of vehicle budget and design difficulty, it would be extremely inefficient. When considering the steering design of a Baja vehicle, it is important to examine three different types of steering geometries: Ackerman, parallel, and reverse. Ackerman steering consists of the inner wheel having a greater turning angle than the outer wheel, reverse Ackerman steering is the opposite, and parallel steering is when both wheels turn at the same angle [4]. To facilitate an Ackerman steering geometry, an Ackerman arm bar design must be used to ensure a tighter turning radius for the inner wheel. For parallel steering, a rack and pinion method would suffice to ensure the same turning radius for each wheel.

A study from the BYU Baja team found that Ackerman steering may be the best method for steering a smaller turn radius and handling static loads [4]. The proposed issue with parallel steering is that the low friction coefficient over a dirt surface will cause a high slip angle similar to that of Ackerman. However, the ability to utilize parallel steering on more rough surfaces results in a minimized slip angle, specifically at higher speeds. While Ackerman steering may be more effective for low speed turning, the priority for high-speed turning and prioritizing a lower slip angle points to the use of parallel steering. The potential for Ackerman steering can be seen more evidently in smaller, lightweight vehicles, specifically those used for the ASME Mini-Baja RC contest. Without the weight of a driver and a minimized cabin, the benefits from a tighter turning radius from Ackerman steering outweigh the potential slipping angle [5]. With the selected parallel steering method and based on the portions of success of previous Union College Baja vehicles, a rack and pinion design will be most efficient and reliable for this year's competition vehicle.

Detailed Design and Alternative Concepts

An overview of the steering rack deign can be broken up into five major subgroups: the rack and pinon, the bearing supports, the key and shaft, the housing around the shaft, and the mount and spacer attachments to the frame. The rack and pinion consist of the rack stock and the gear which will be inputting the torque on the rack. The bearing supports will be on either side of the gear, as well as inside the housing to allow for minimal friction and smooth travel. The key allows for the transmission of the torque from the steering column to the rack. The key comes into contact with the gear and the key shaft, which is all surrounded by the housing. The housing is where most of the issues resided in last year's design, so solidifying the housing was a major emphasis during the design process. The mount is the portion of the rack which is attached to the frame of the vehicle and endures the most load. A spacer was added between the mount and the housing to increase the weld points on the frame from the mount, thus increasing the factor of safety.

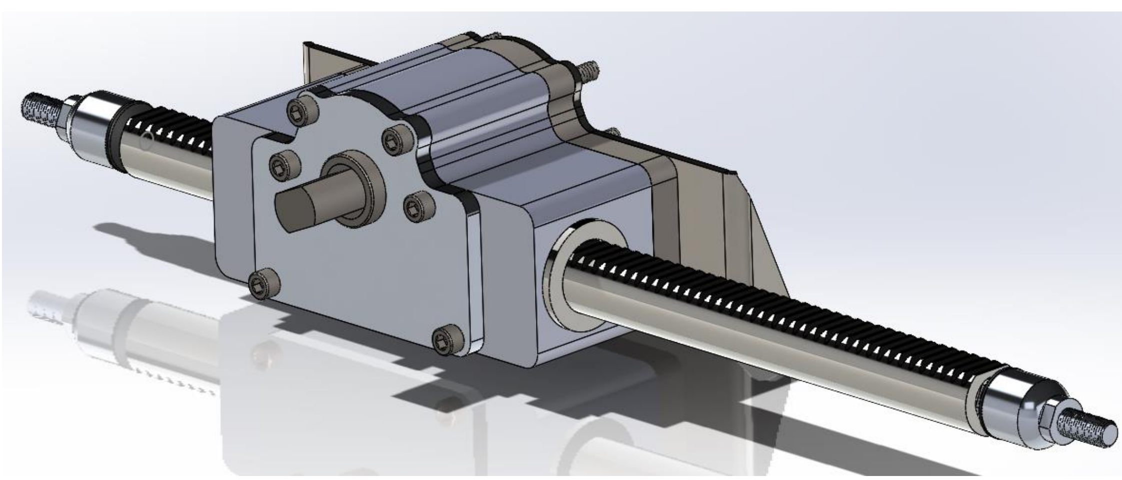


Figure 1: Full assembly of the steering rack system 2022-2023.

Rack and Pinion:

The selection of the rack and pinion was made based on the consideration of two design requirements: the travel distance and the wheel angle. The travel distance of 1.5 in. corresponds directly to the gear ratio and wheel angle during this selection process. Last year, a rectangular rack was chosen with a 0.5" length. For better rotation and travel along the rack, a circular rack was chosen instead of rectangular. This also minimizes the friction during travel. To increase the strength of the rack, a 1" diameter rack was chosen, doubling the size from last year. The pressure angle for the rack is 20°, with a pitch of 16, both of which need to be matched by the gear. For the gear, a smaller pitch diameter was chosen than in years past. Based on the 180° of travel which was desired by the driver, the larger gear used in years past (1.5" PD) would only allow for 114.6° of rotation with the full 1.5 inches of travel, and only 76.4° with 1 inch of travel. A gear pitch diameter of 1" gives us the best steering wheel angle at 171.9° with the full 1.5 inches of travel. It was then decided to use the model with hardened teeth. This is due to the behavior of teeth when under stress. The teeth are the first part of the gear which fail when put under maximum loading.

Figure 2: Hardened teeth metal gear: 1" pitch diameter, provided by McMaster Carr.

Because of a smaller gear pitch diameter, it was necessary to confirm that the teeth of the gear would not fail under the torque applied from the shaft. For this calculation, it was assumed that the typical maximum force applied by a driver was roughly 30 pounds. For this simulation, the force was doubled to assume that the driver exerts this maximum fore with both hands. Using the Lewis Factor of Safety Equations, and the carbon steel material properties of the gear, the factor of safety was found to be 5.26. The smaller pitch diameter seemed to have very little effect on the strength of the teeth.

Bearings and Supports

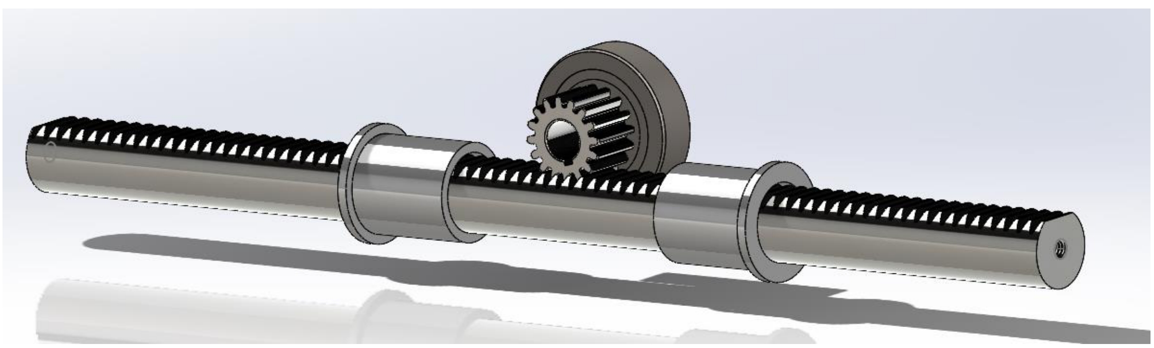


Figure 3: Assembly containing the rack and pinion, with the high load ball bearing and the flanged bronze oil embedded bearing.

To secure the rear hub of the gear, and translate the torque from the shaft, a high load ball bearing was placed around the hub of the gear. It is important to note that the inner diameter of the ball bearing does not match the outer diameter of the hub. Therefore, modifications are required to machine the outer diameter of the hub from 0.813” to match the inner diameter of the hub at 0.75” to be press-fit. Figure 3 shows the ball bearing placed on the hub on the back of the gear, as well as two bearings. On the left and right of Figure 3 are Flanged bronze Oil Embedded bearings,

which will rest on rack to allow for minimal friction and smooth motion. The inner radius of the bearings matches the 1” diameter of the rack and will rest inside the housing. The bearings will be press fit into the housing to eliminate axial movement not induced by the torque of the shaft.

Key and Shaft

For the torque to be transmitted from the shaft to the gear and initiate motion of the rack, a key is needed. This key is placed in a shaft, where half of the key is in contact with the gear, and half is in contact with the shaft. Figure 2 shows the key cut in the gear with dimensions of 0.25” x 0.25” x 1.25”. The use of a key prevents constant rotation of the shaft when torque is applied through the steering column. Since the key is responsible for the motion of the gear, which directly facilitates the motion of the rack, a stress test was necessary to ensure that the key would not fail under the shear stress and bearing stress. The same torque which was used for the stress on the gear teeth was used for the key. The sheer stress was found to exert 4224 Psi, resulting in a factor of safety of 5.21 proving that the key was plenty strong enough to not fail under maximum torque. The shaft will be cut from 4140 alloy steel, with an inner diameter of 0.5” to match the gear, and a shoulder with a diameter of 0.75”, to prevent axial movement of the shaft. A cut was made in the key shaft, as shown in Figure 4 for the key. A groove was also cut into the middle of the shaft for a snap ring to prevent axial movement within the steering system.

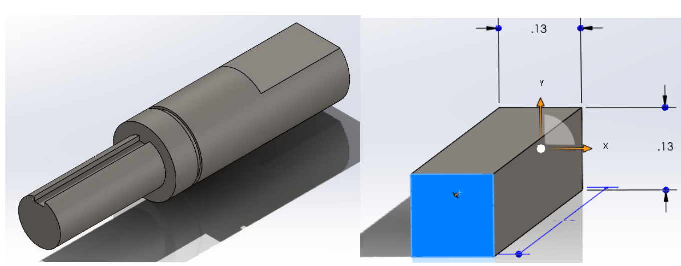


Figure 4: 1045 Carbon Steel key and a 4140 Alloy Steel Key Shaft with a key cut.

On the other end of the shaft, an input is cut in an oval shape to fit into the steering column, eventually leading to the steering wheel. This is to fit inside of the column that will receive the input torque from the driver. The steering column is designed by another member of the Baja team, emphasizing the importance of communication and connection between different subsystems of the vehicle.

Housing:

The housing of the steering rack is arguably the most important part of this subsystem. The housing is where most issues arose from last year's design, which resulted in the initial 2021 rack not being usable for the car. The housing must be able to withstand torque and static loads, while being low weight and efficient. The housing is the area which must be constrained the most to fit within the entire vehicle assembly.

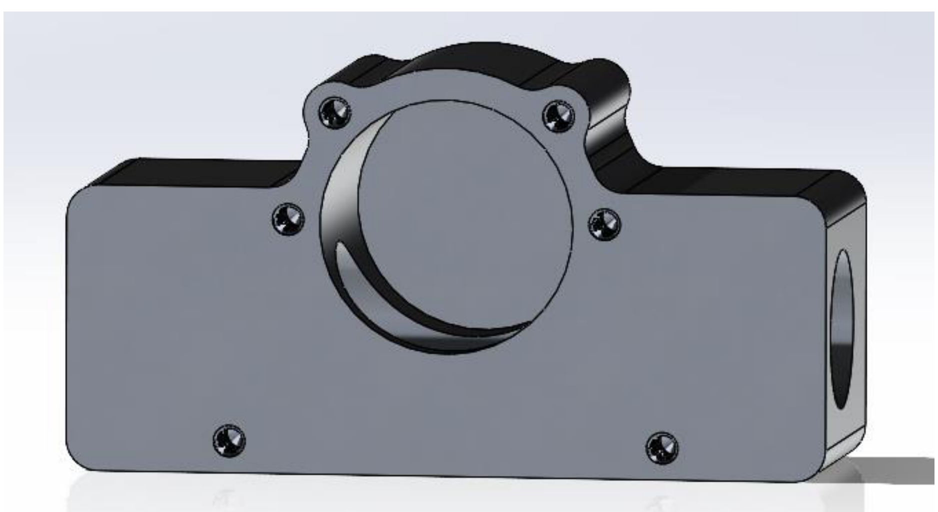


Figure 5: Middle housing made from 6061 Aluminum block

Figure 5 shows the design for the middle housing piece. For last year's design, the housing was split up into three parts: a front cap, a middle section, and a back plate. This middle housing would encompass the middle and back plate from last year's design. The entire housing is 6 in. long and 2.19 in. in width. There is a 1.25-in hole which cuts through the entire housing to hold the rack. The 1.25-in diameter is to hold both the rack and the oil-embedded bearing supports. The housing sits on these bearings to allow for smooth horizontal movement and minimal friction. The middle circular cut has a diameter of 2 in. to hold the outer diameter of the ball bearing to be press fit inside the housing. The wall behind the ball bearing is 0.125 in. thick. The entire block is made from 6061 Aluminum, with 6 tapped holes for 0.25-in screws. The fillets which are around the holes, and sides of the housing have a radius of 0.25 in. for smoothness and safety. The screws will be drilled into a front plate for support and stability of the entire housing.

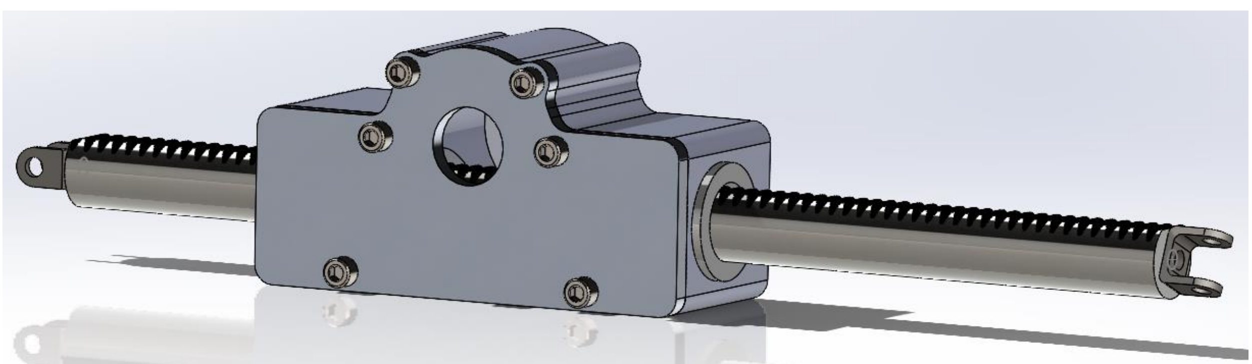


Figure 6: Housing mounted onto rack with front plate and screws

As shown in Figure 6, the front plate acts as a support and stability plate for the entire housing. The front plate is 0.25 in. thick, and matches the same geometry as the middle housing, aside from the hole cut. The hole cut matches a support bushing that will encounter the shaft, to

be addressed in the final assembly. The 6 screws are placed around the gears and rack to maximize stability.

For manufacturing, the thickness of the middle housing requires it to be created using a computerized numerical control (CNC) machine, to be cut from a block of 6061 aluminum. The front plate is very thin, and the geometry will allow it to be cut using a water jet. The water jet machine is extremely efficient and accurate.

Mount:

Following the design of the steering rack, the next vital part of the assembly was the mounting system. The mounting system is designed to attach the bottom of the housing to the frame of the vehicle, securing it in place. An important aspect to note about the mount is the potential load which it endures when coming into contact with obstacles or other vehicles. The torque applied to the rack when the vehicle is fully turned could potentially torque the entire system rather than only transmit a torque on the shaft and gear. In last year's model, the whole steering system began to torque when a heavy load was applied. Figure 7 shows the mount design within the vehicle frame which was chosen for this year's system. Initially, the mount was raised higher than what is shown in Figure 7, and the only points for welding were on the left and right ends. After FEA analysis of this arrangement, the system failed with a load of 1100 lbf. This load is to simulate a crash when the vehicle is fully turned. Following further discussion and recommendations from different team members, the mount was lowered, as shown in Figure 7, so that the weld point could be across the entire bottom layer of the mount. This gives a sturdier base, and resulted in a successful FEA test, to be discussed in the next section. To keep the steering rack in its current position and be able to lower the mount, a spacer was added as a placeholder between

the two parts, as shown in Figure 8. The cut of the spacer matches the top of the mount, for a perfect fit.

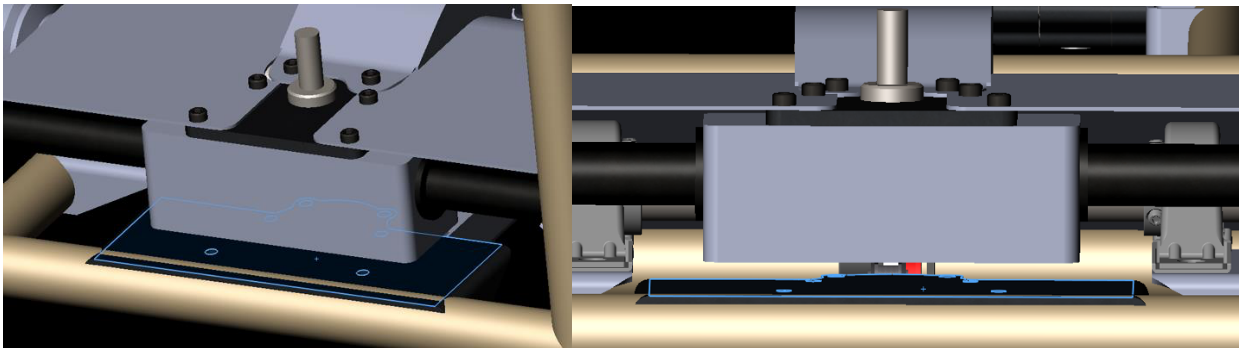


Figure 7: Steering rack and mount within vehicle frame.

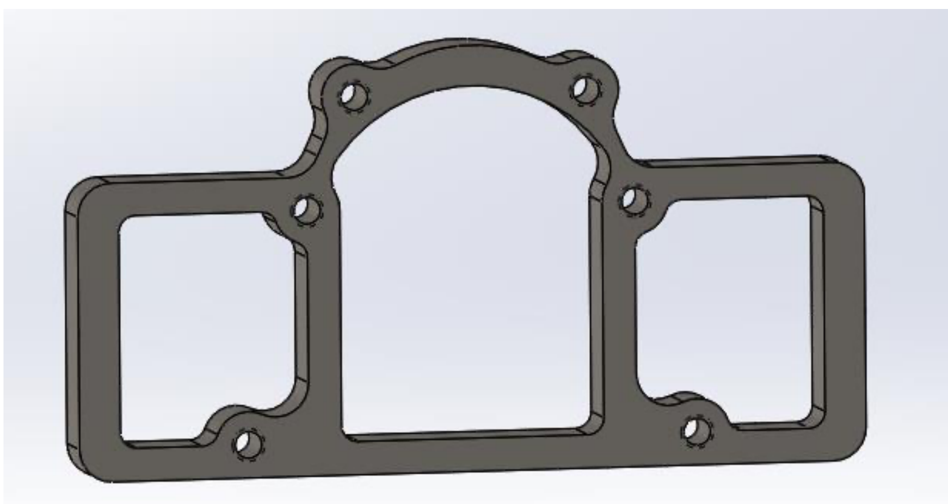


Figure 8: Spacer between the steering rack and mount.

Assembly Overview:

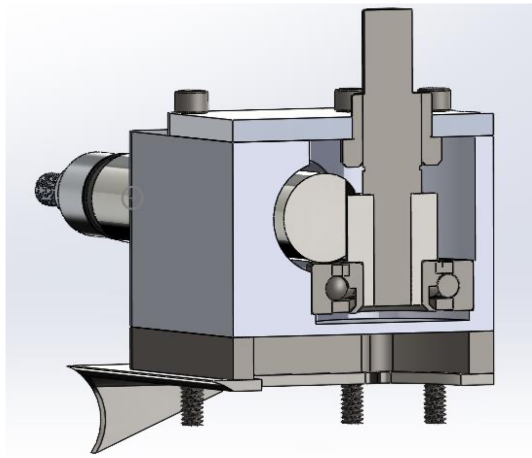


Figure 9: Full assembly cross section cut showing bearings and supports

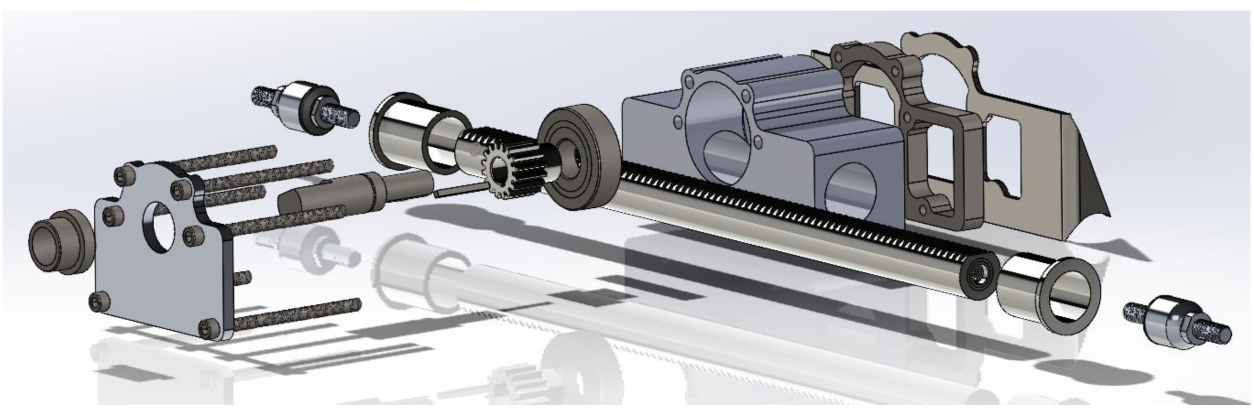


Figure 10: Full assembly exploded view

Figures 9 and 10 show an overview of the entire steering rack system. The support system is displayed more in depth in Figure 9. It was important to consider how the shaft was going to be supported while inside the steering rack. If axial movement occurred, the shaft would slide right out once any force was applied. If there were vertical movement allowed within the shaft, the

bending moment would cause possible failure within the steering system. The press fit drill bushing solves the vertical movement issue, as well as half of the axial movement concerns. The bushing surrounds the outer diameter of the shaft exiting the front plate to secure it in place. The bushing is press fit into the front plate for stability. The shoulder of the shaft also prevents axial movement. The diameter of the shoulder is 0.25” larger than the rest of the shaft diameter. The shoulder rests between the gear and the bushing, preventing any possible axial movement.

The exploded view gives great insight into the full assembly of the steering rack in Figure 10. The key is shown between the gear and the key shaft, fitting perfectly into the cut on each part. The ball bearing is press fit into the housing, and the inner diameter of the ball bearing fits directly around the hub of the gear. The six screws secure the front plate to the middle housing and provide support and stability for the rack. Shown on the left and right of the rack in Figure 10 are the inline ball joint linkages. These allow for the connection of the rack to the suspension system. To connect the linkage to the rack, a hole is drilled into the rack for maximum strength. The through bolts will go into the spacer, where the holes are threaded. The spacer will then be welded to the mount for maximum support.

FEA:

While the steering rack design may fit geometrically into the vehicle assembly, further analysis must be done to ensure the reliability and efficiency of the rack. The Lewis force and load equations have been calculated to determine the factor of safety for the gear teeth and the key. This is not the only area which a load is applied onto the rack. Another area which brings about a static load is a collision. If the vehicle meets another vehicle head-on, or even an obstacle, a load is applied to the front of the wheels. If the wheels are turned during this collision, a load is applied

on the rack. If the rack fails, the tires will forcefully turn in the direction of the applied force, and the steering system will fail entirely.

It is difficult to quantify the exact force which will occur during a collision. This is due to the uncertainty of the angle which the obstacle will encounter the vehicle, the angle which the wheels are set at the instant of collision, and the angle of suspension which is subject to change ±17 °. Another aspect of uncertainty which makes this analysis difficult to quantify is the area on which the load is applied in the steering rack. When looking at Figures 7 and 8, we can see that a load would be applied at the clevis connection points and transmitted directly into the rack. The suspension is normally at 8 ° ± 17 ° below horizontal, but to maximize safety we assume the entire load is being applied at horizontal. In discussion with other Baja chiefs and based on prior experience, a safe estimate for an applied force is three times gravity. This was deemed to account for more than enough safety when on the test track. While this load is being applied directly at horizontal, it is not just transmitted to the rack. The rack will transmit the load to the gear, which will translate this load to the hub on the back of the gear. The hub transmits this load to the ball bearing on the back of the hub. The ball bearing is press fit into the middle housing. The various steps in load transmission and uncertainty of collision force makes this analysis so difficult to quantify. Finite element analysis (FEA) was used on SolidWorks to better represent this scenario. For maximum safety purposes, the entire 3g load was applied on the inside face of the middle housing, which contains the press-fit ball bearing.

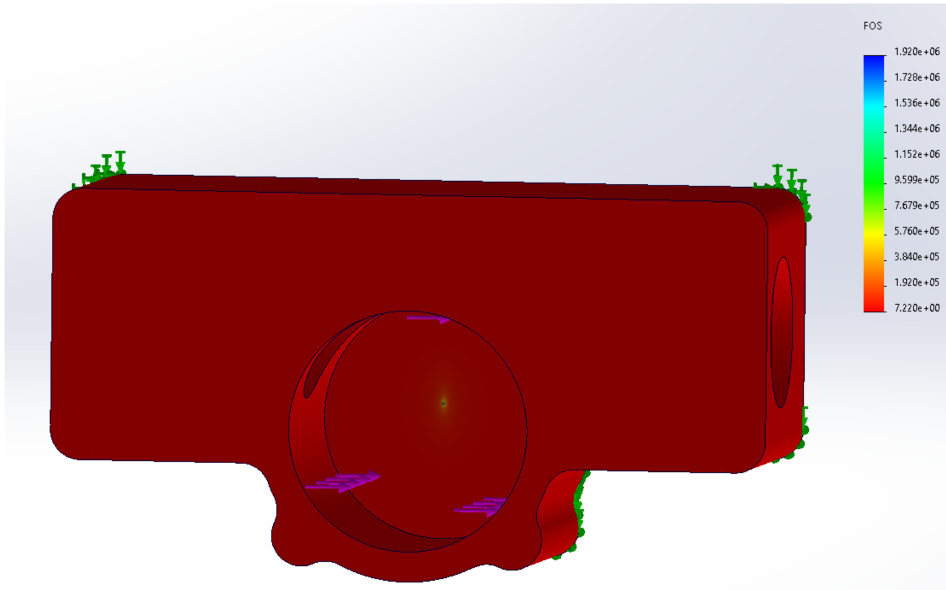


Figure 11: FEA analysis of the middle housing, showing an average factor of safety of 7.2.

Figure 11 shows the results from the finite element analysis of the middle housing, displaying a factor of safety of roughly 7.2, which displays a large level of confidence for collisions and obstacles. The back of the middle housing was fixed, to simulate the mounting system on the vehicle assembly. The reality of the middle housing undertaking the full 3g load is very much an overestimate, and this large factor of safety gives room for edits within the design.

Similarly, with the key shaft, FEA analysis was completed to determine if the key shaft could withstand the torque from the steering column. The key shaft transmits the torque from the steering wheel to the key, which is then transmitted to the gear. The factor of safety on the key was already shown using the Lewis equations, but the key shaft also had to withstand this load. Like the key, the load was determined from a 30-pound force applied to the steering wheel per hand. This torque was inputted on the FEA analysis and the results are shown in Figure 12.

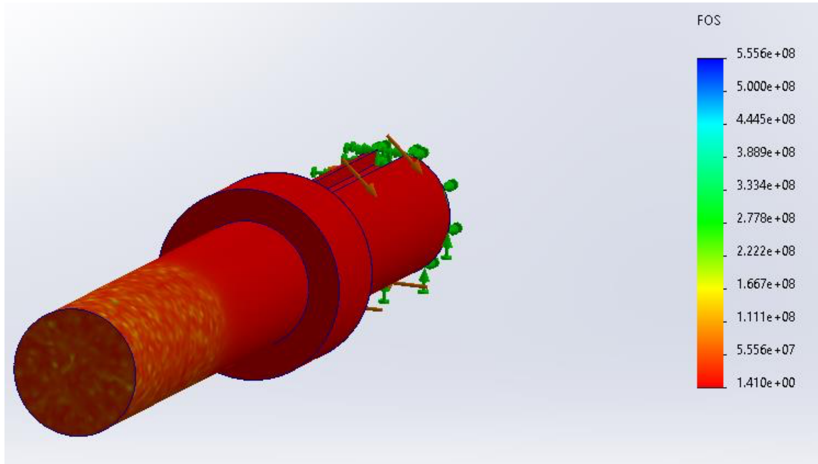


Figure 12: FEA analysis of the key shaft, showing an average factor of safety of 1.4

As shown in Figure 12, the average factor of safety displayed on the entire key shaft is 1.4. While this would normally be a low factor of safety on any material, but the torque which was inputted on the shaft was much higher than what would be realistic for a driver. Normally, a driver exerts a maximum force of 30 pounds on the wheel, but for this study, the force was doubled for both hands. The FEA analysis for both aspects of the steering rack has increased confidence for the efficiency and reliability for the entire rack.

Following the FEA of the shaft was the FEA done on the mount. This was arguably the most important FEA analysis of the system, due to the load transmitted to the entire mount. As stated before, the initial mount had failed due to lack of support on each end. Figures 13-14 show the final FEA analysis of the mount, resulting in a factor of safety of 1.9, and negligible displacement. As shown in Figure 14, with the newly modified support of the mount, the point of maximum stress was no longer in the mount but was inside the housing. This gives the utmost confidence in the reliability and efficiency of the steering rack.

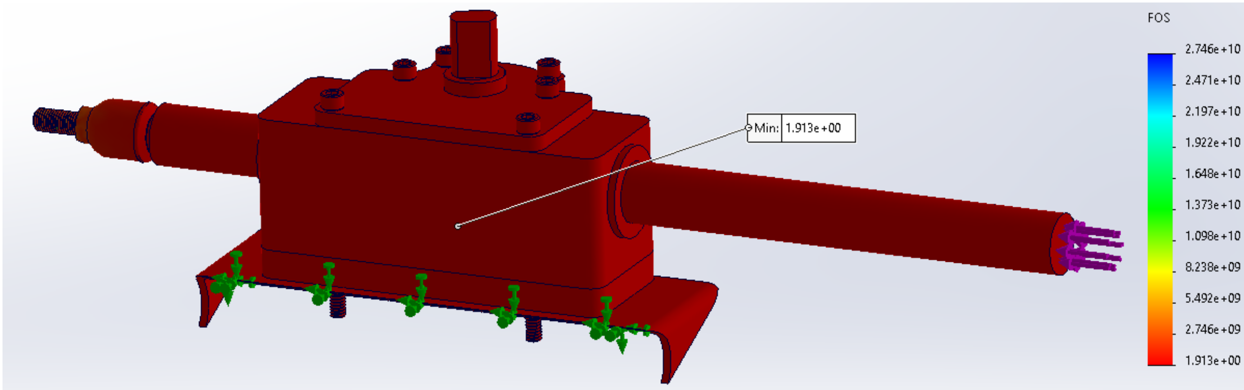


Figure 13: FEA analysis of the entire system showing a factor of safety of 1.9

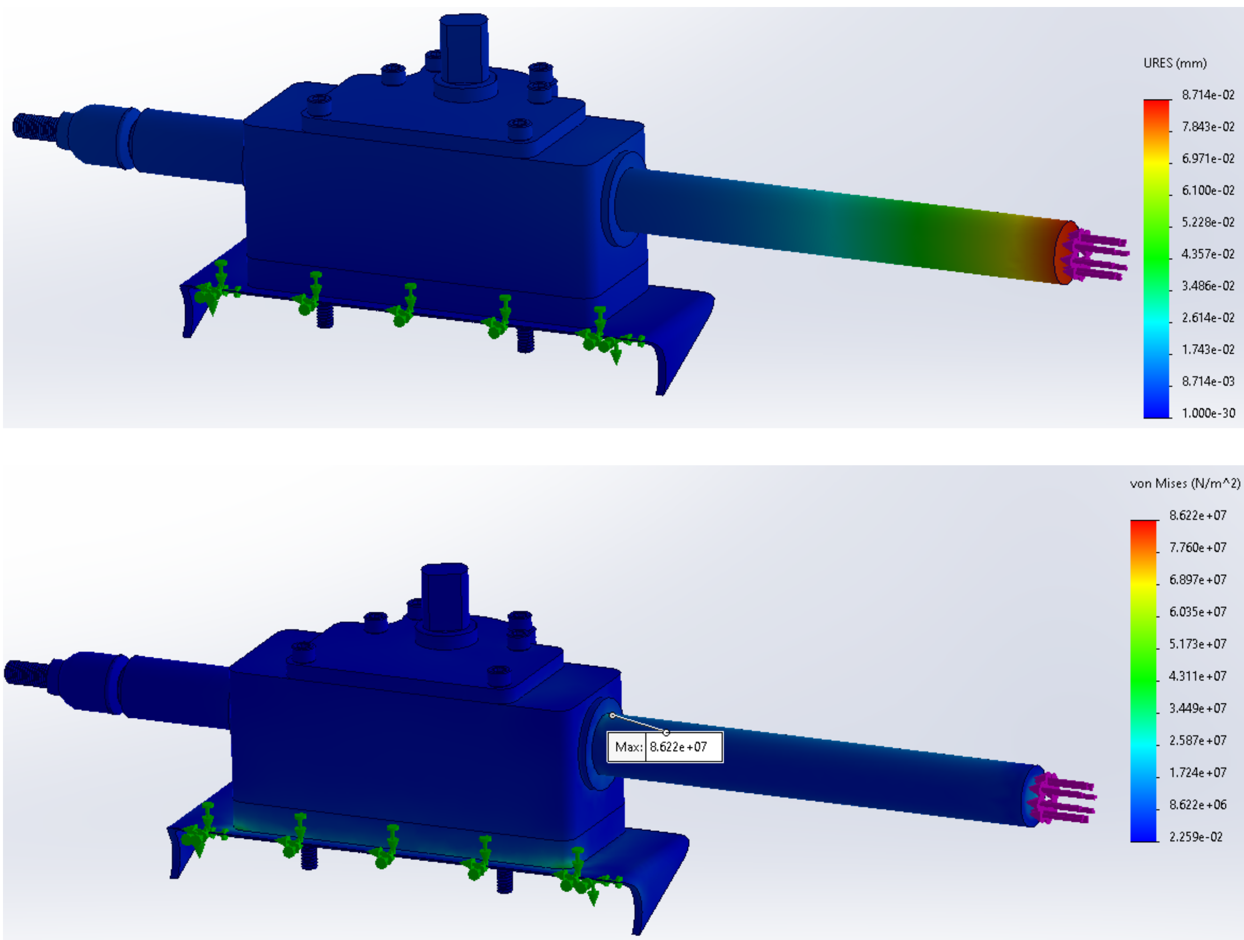


Figure 14: FEA analysis of the entire system showing negligible displacement, and a maximum stress inside the housing of the steering system

Modifications:

There have been several modifications within the steering rack, leading to the final design in its current state, one of the most important being the middle housing. Figure 15 shows the old middle housing compared to the current design. The old middle housing is bigger, stronger, and more reliable, but when placed into the large vehicle assembly, there was not much room between the drive train sprockets and the back of the middle housing. The new middle housing was created and saved 0.5 inches in thickness, as well as an extra 0.25 inches from the back wall of the housing behind the ball bearing. While these were large portions of the rack which were removed, the factor of safety for the middle housing is well above what is necessary for operation. Another modification came through the key shaft. Communication within the different Baja systems is key for smooth operation. The chief of cabin helped to determine the proper radius for the outside of the key shaft, so that it may properly fit into the steering column.

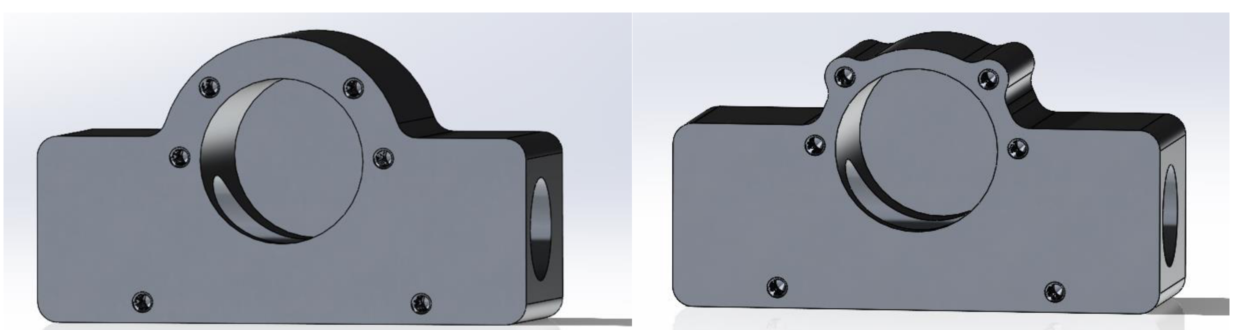


Figure 15: Comparison of the middle housing designs, before and after changes in geometry.

The front plate has come subject to modification as well. The front plate is meant to match the geometry of the middle housing. As the middle housing changes, so does the front plate. The front plate will also be trimmed from the outside of the bolts. The area to the left and right of the

bottom bolts only adds weight, and further modification of this front plate can help to make the steering rack more efficient. Lastly, the inline ball joint linkages were chosen as opposed to the original clevis designs in term 1. This came at the recommendation of the chief of suspension, providing more movement and stability to the steering system as it connects to the suspension.

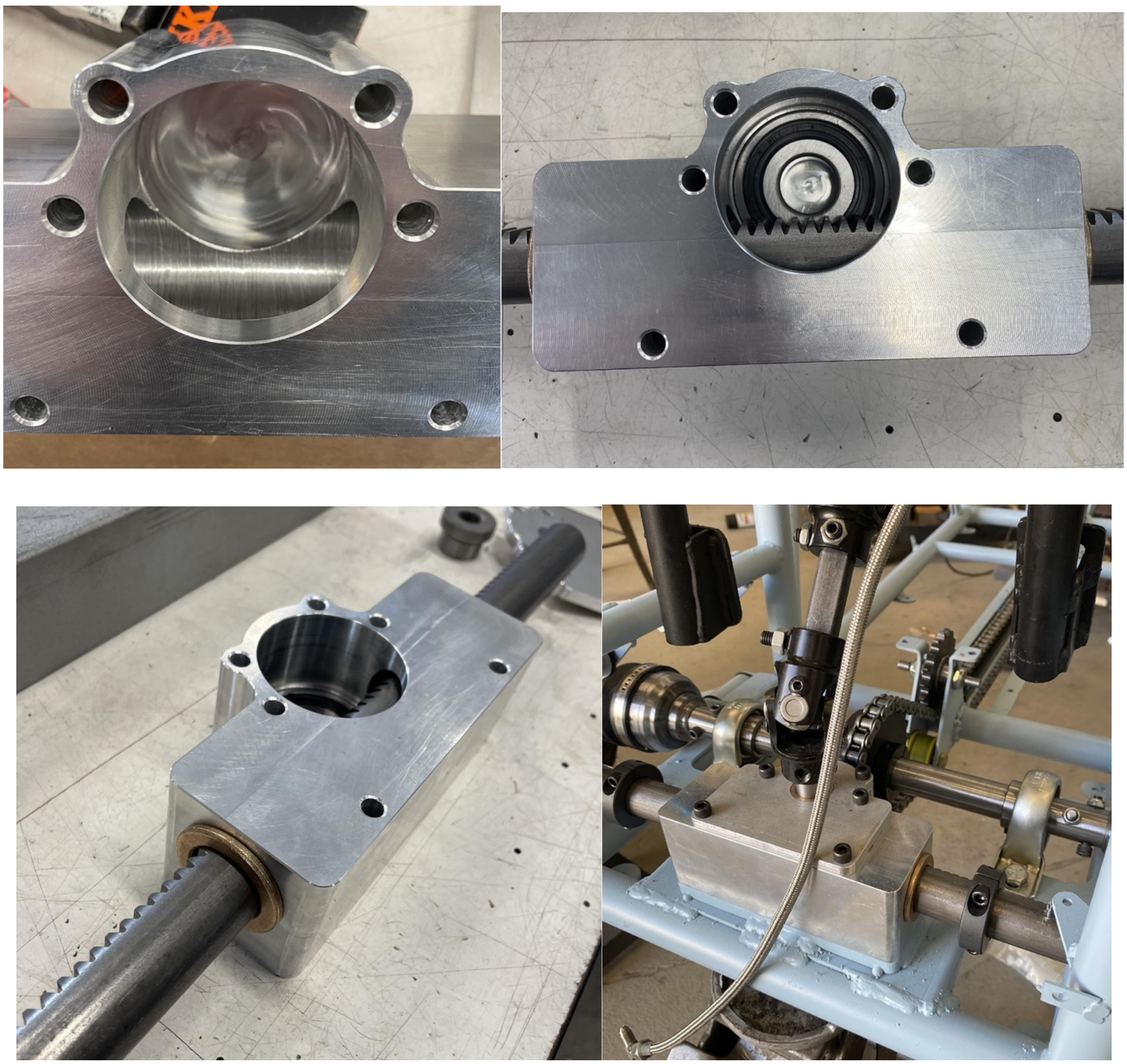


Figure 16: Images of the manufacturing process of the steering system

Performance Review:

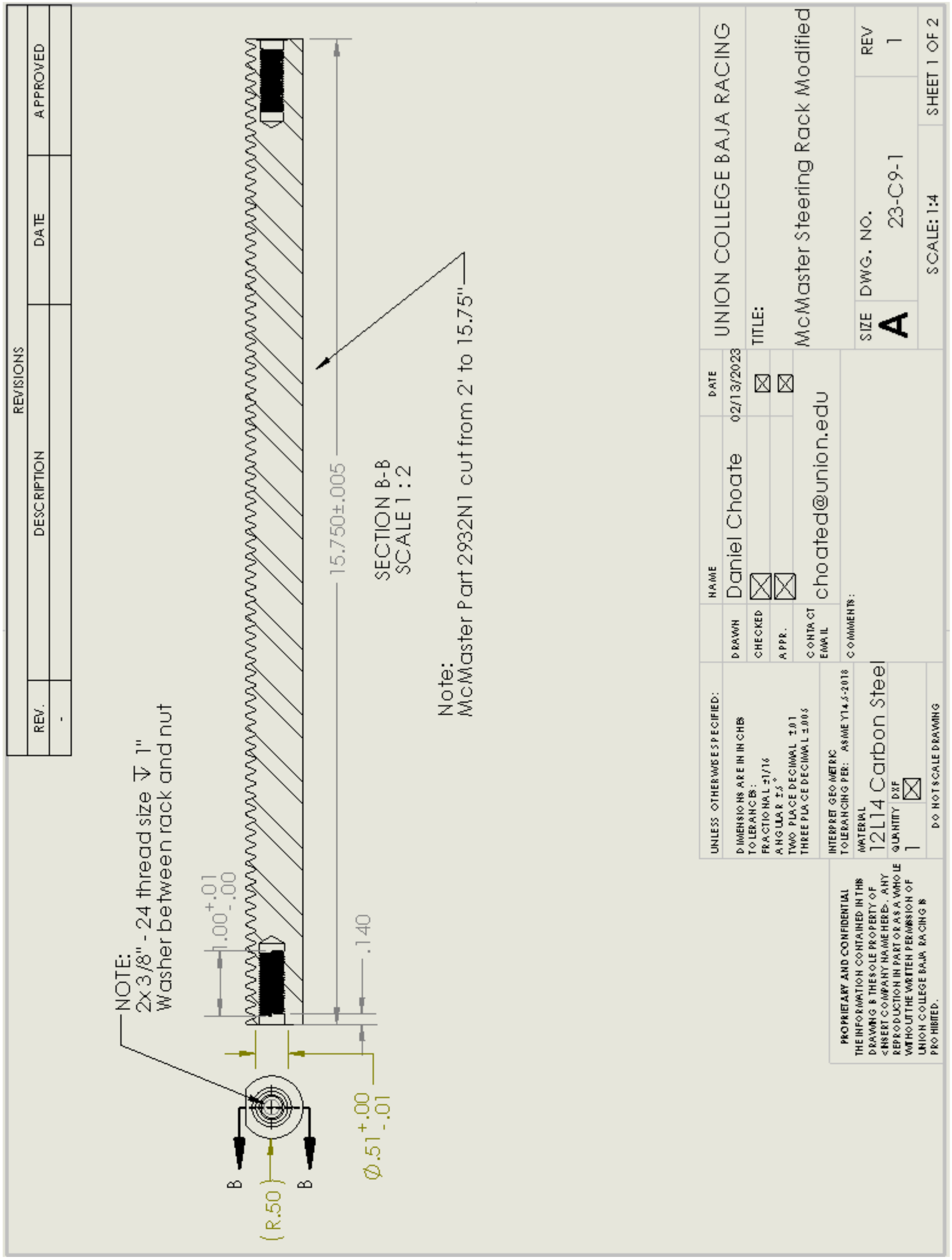
Following the manufacturing process and implementation into the vehicle, the steering system was put to the test at this years' competition in Oshkosh, WI. Based on the tech inspections, and endurance test performance, the steering system proved quite reliable and effective in terms of being able to withstand extreme loads and collisions. One area of improvement which may be focused on for next year's vehicle would be the turning radius. The turning radius was not as sharp as desired, and therefore resulted in the vehicle being unable to maneuver through cones on a tight obstacle course.

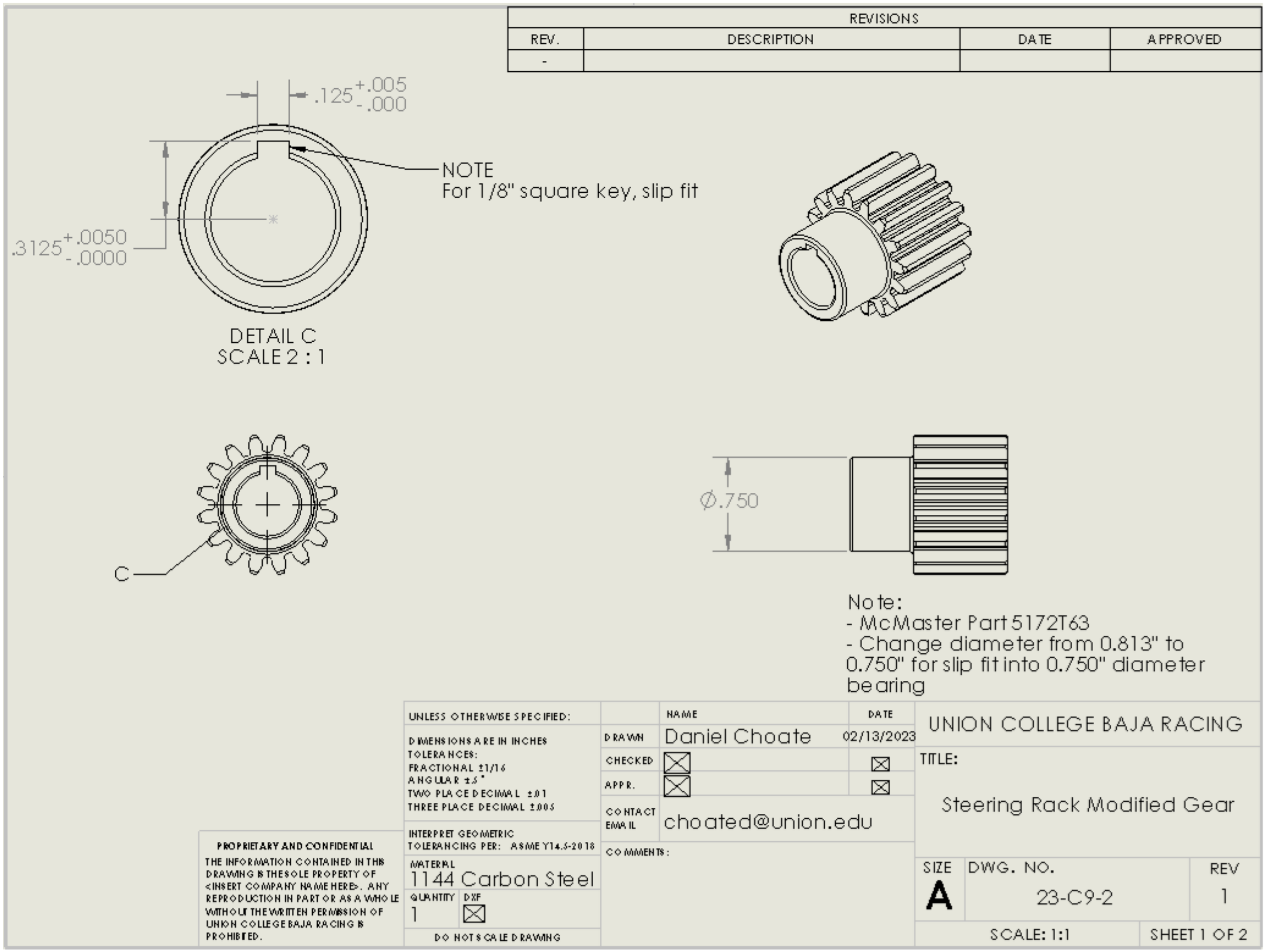
Overall, this design has not only been impactful for this year's vehicle and performance but will also serve as a standing model for future Union College teams and vehicles. I final score of 208.41 was quite impressive, considering the size and resources of a smaller scale team at Union College. Through future iterations of the vehicle in years to come, the performance and success of the Union College SAE Baja Team will continue to rise.

Acknowledgements:

I would like to acknowledge the following people who have been instrumental in the design, manufacturing, and guidance of this research exploration: Professor William Keat, Paul Tompkins, Robert Harlan, and Professor Andrew Rapoff.

Appendix A: Dimensioned Drawings




REVISIONS
REV.
DESCRIPTION
DATE
APPROVED
-
.125 +.005 -.000
NOTE
For 1/8" square key, slip fit
.3125 +.0050 -.0000
DETAIL C
SCALE 2 : 1
C
Ø.750
Note:
- McMaster Part 5172T63
- Change diameter from 0.813" to 0.750" for slip fit into 0.750" diameter bearing
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±1/16
ANGULAR ±.5°
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005
INTERPRET GEOMETRIC TOLERANCING PER: ASME Y14.5-2018
MATERIAL
1144 Carbon Steel
QUANTITY
1
DXF
DO NOT SCALE DRAWING
PROPRIETARY AND CONFIDENTIAL
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NAME
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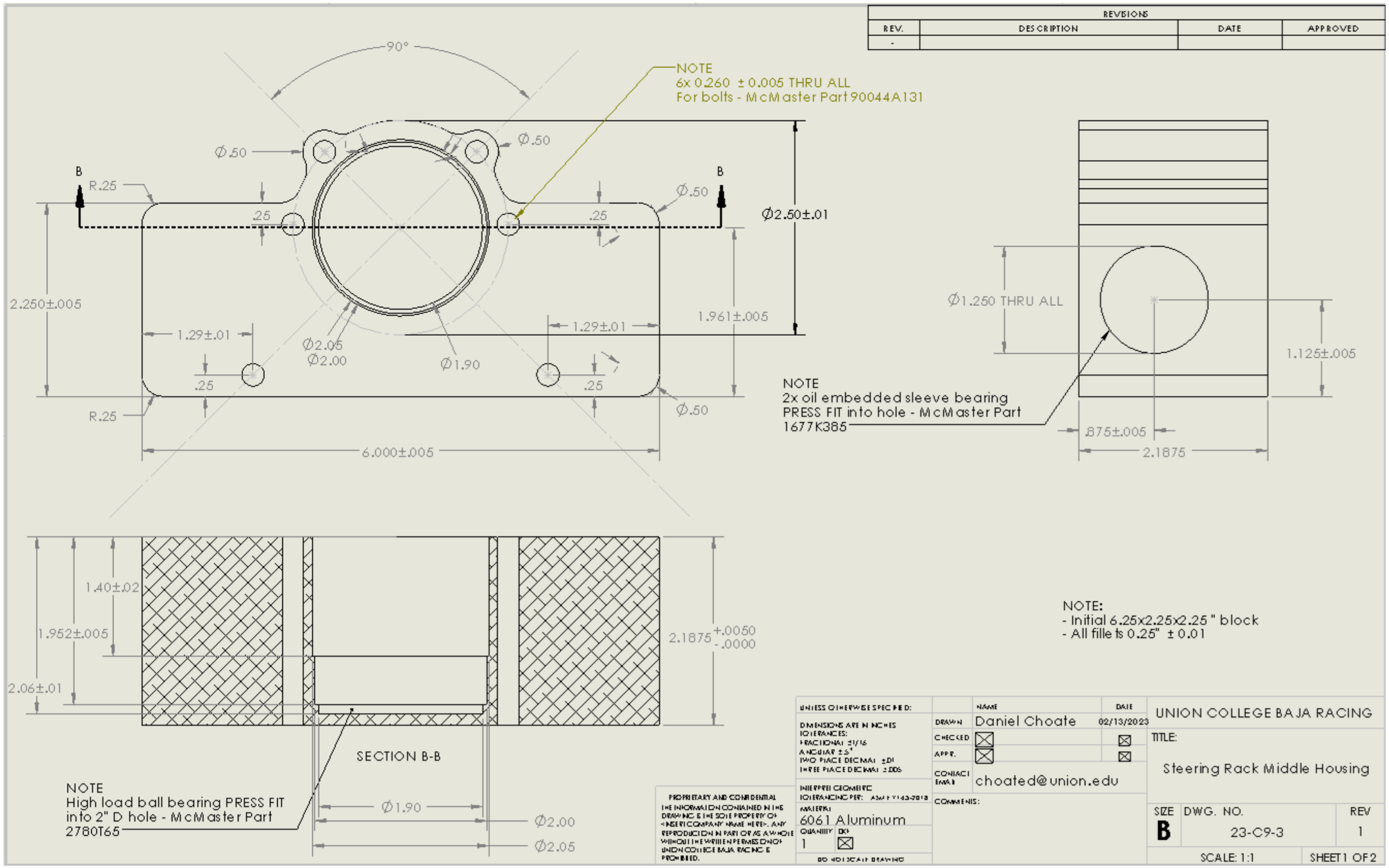
DRAWN
Daniel Choate
02/13/2023
CHECKED
APPR.
CONTACT EMAIL
choated@union.edu
COMMENTS:
UNION COLLEGE BAJA RACING
TITLE:
Steering Rack Modified Gear
SIZE
A
DWG. NO.
23-C9-2
REV
1
SCALE: 1:1
SHEET 1 OF 2


REVISIONS
REV.
DESCRIPTION
DATE
APPROVED
NOTE
6x 0.260 ± 0.005 THRU ALL
For bolts - McMaster Part 90044A131
90°
Ø.50
Ø.50
Ø.50
Ø.50
R.25
R.25
.25
.25
.25
.25
Ø2.50±.01
2.250±.005
1.29±.01
1.29±.01
Ø2.05
Ø2.00
Ø1.90
1.961±.005
6.000±.005
Ø1.250 THRU ALL
1.125±.005
NOTE
2x oil embedded sleeve bearing
PRESS FIT into hole - McMaster Part
1677K385
.875±.005
2.1875
1.40±.02
1.952±.005
2.06±.01
2.1875 +.0050 -.0000
SECTION B-B
Ø1.90
Ø2.00
Ø2.05
NOTE
High load ball bearing PRESS FIT
into 2" D hole - McMaster Part
2780T65
NOTE:
- Initial 6.25x2.25x2.25" block
- All fillets 0.25" ± 0.01
UNLESS OTHERWISE SPECIFIED:
NAME
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02/13/2023
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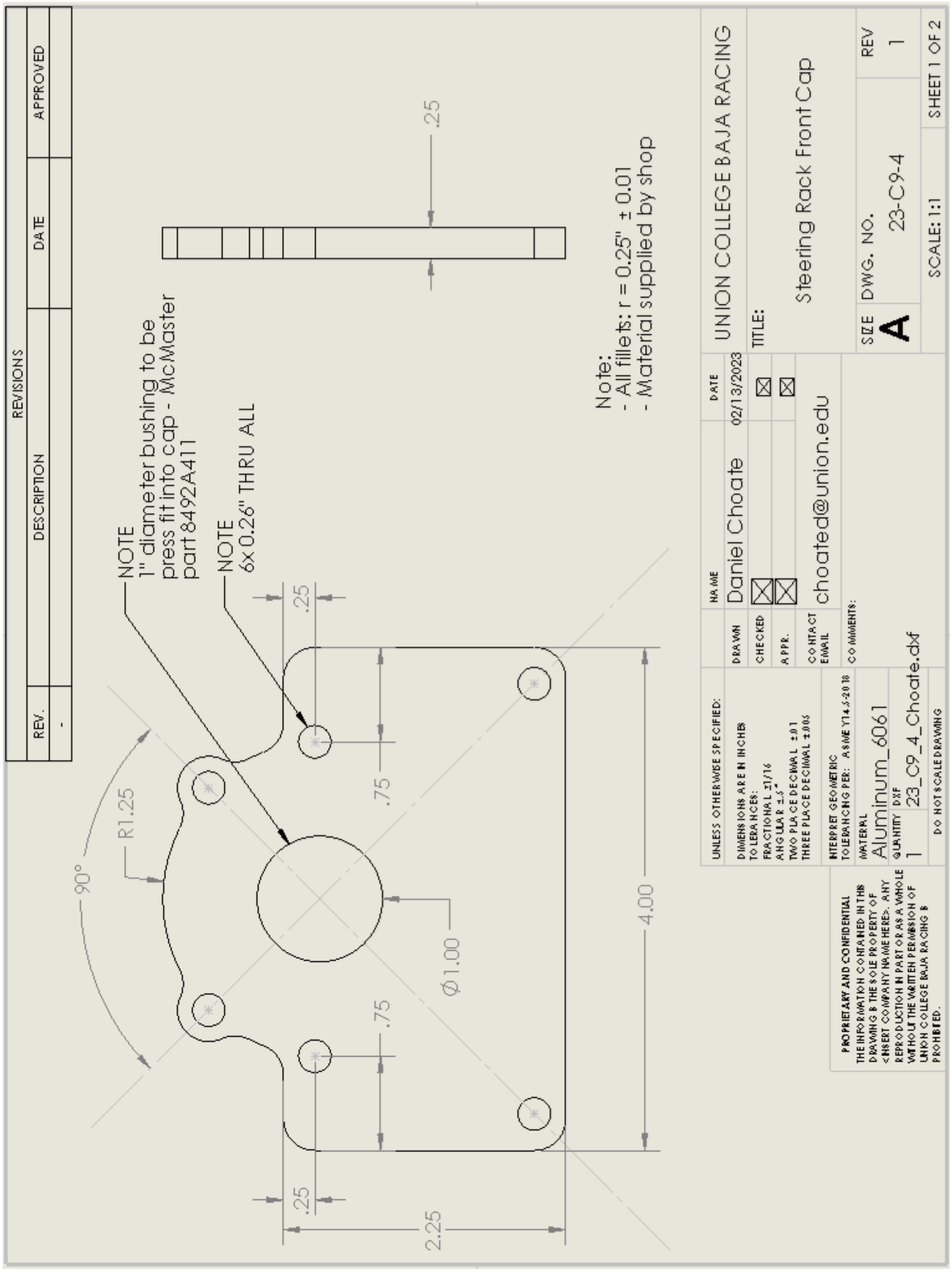
CONTACT EMAIL
choated@union.edu
COMMENTS:
MATERIAL
6061 Aluminum
QUANTITY
1
DO NOT SCALE DRAWING
UNION COLLEGE BAJA RACING
TITLE:
Steering Rack Middle Housing
SIZE
B
DWG. NO.
23-C9-3
REV
1
SCALE: 1:1
SHEET 1 OF 2


REVISIONS
REV.
DESCRIPTION
DATE
APPROVED
-
NOTE
1" diameter bushing to be press fit into cap - McMaster part 8492A411
NOTE
6x 0.26" THRU ALL
90°
R1.25
Ø1.00
.75
.75
.25
.25
.25
4.00
2.25
Note:
- All fillets: r = 0.25" ± 0.01
- Material supplied by shop
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±1/16
ANGULAR ±.5°
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005
INTERPRET GEOMETRIC TOLERANCING PER: ASME Y14.5-2018
MATERIAL
Aluminum_6061
QUANTITY
1
DXF
23_C9_4_Choate.dxf
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Daniel Choate
02/13/2023
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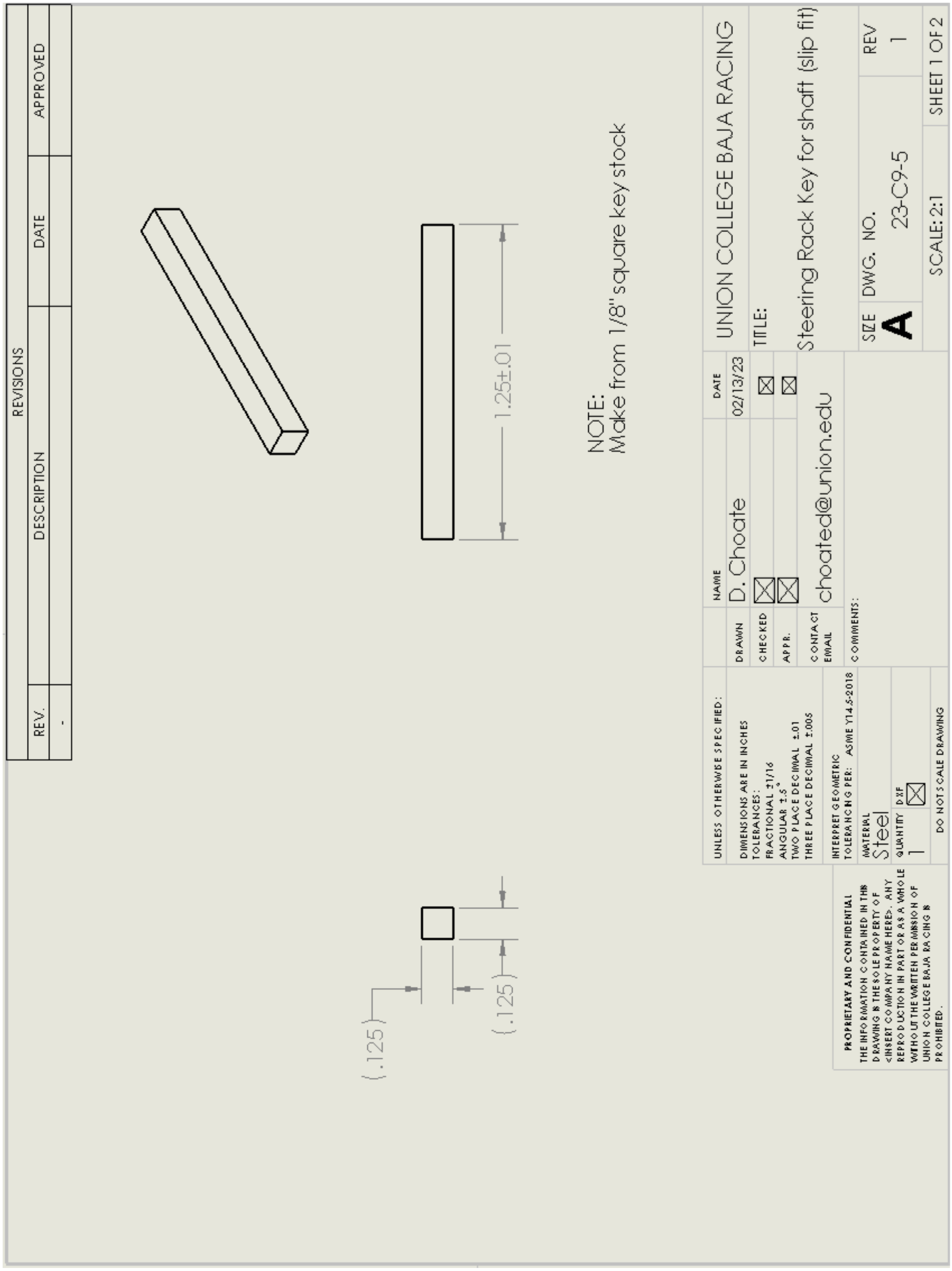
CONTACT EMAIL
choated@union.edu
COMMENTS:
UNION COLLEGE BAJA RACING
TITLE:
Steering Rack Front Cap
SIZE
A
DWG. NO.
23-C9-4
REV
1
SCALE: 1:1
SHEET 1 OF 2


REVISIONS
REV.
DESCRIPTION
DATE
APPROVED
-
(.125)
(.125)
1.25±.01
NOTE:
Make from 1/8" square key stock
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±1/16
ANGULAR ±.5°
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005
INTERPRET GEOMETRIC
TOLERANCING PER: ASME Y14.5-2018
MATERIAL
Steel
QUANTITY
1
DXF
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02/13/23
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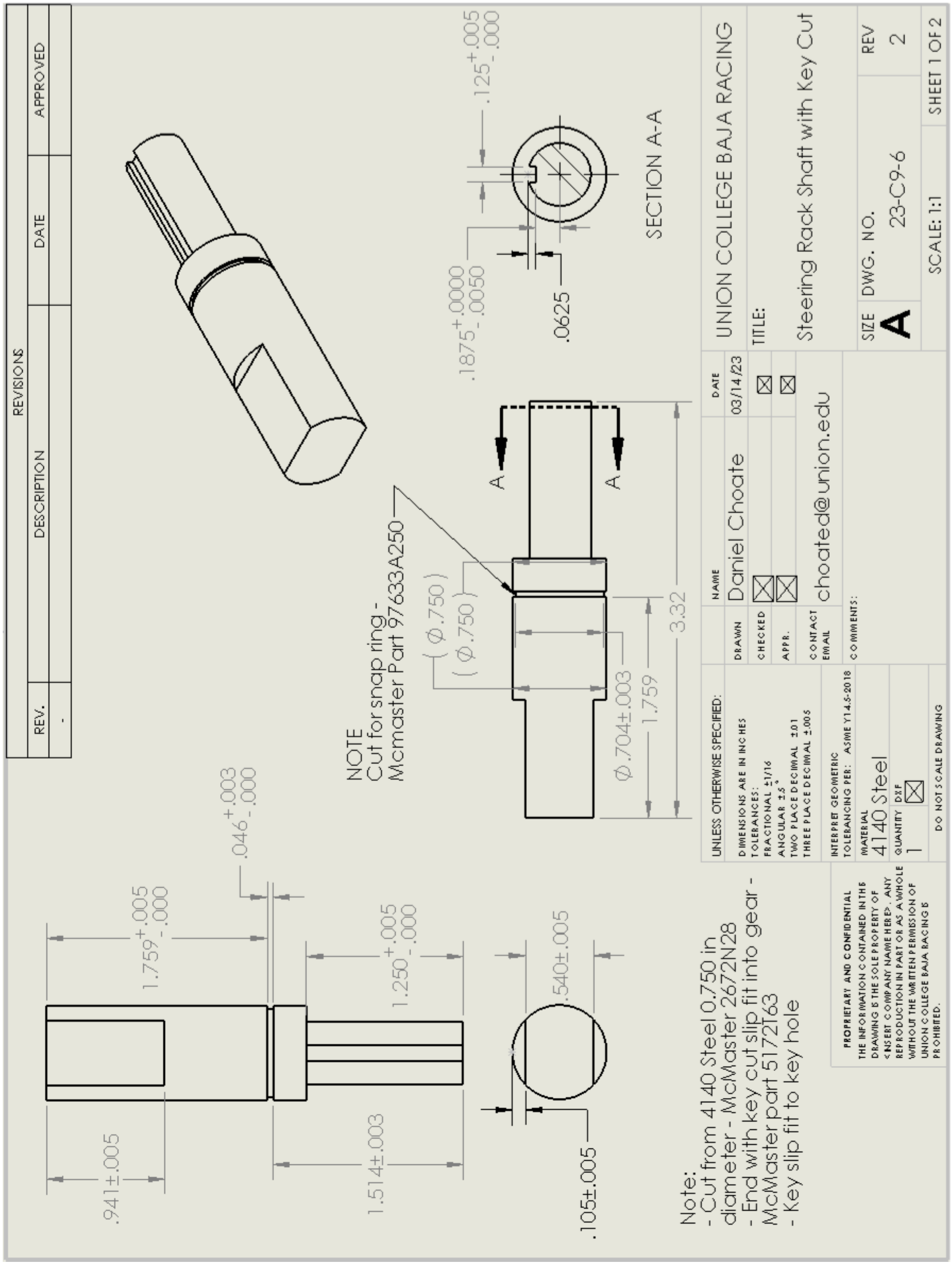
CONTACT EMAIL
choated@union.edu
COMMENTS:
UNION COLLEGE BAJA RACING
TITLE:
Steering Rack Key for shaft (slip fit)
SIZE
A
DWG. NO.
23-C9-5
REV
1
SCALE: 2:1
SHEET 1 OF 2


REVISIONS
REV.
DESCRIPTION
DATE
APPROVED
-
.941±.005
1.759+.005 -.000
.046+.003 -.000
1.514±.003
1.250+.005 -.000
.540±.005
.105±.005
Note:
- Cut from 4140 Steel 0.750 in diameter - McMaster 2672N28
- End with key cut slip fit into gear - McMaster part 5172T63
- Key slip fit to key hole
NOTE
Cut for snap ring -
Mcmaster Part 97633A250
(Ø.750)
(Ø.750)
Ø.704±.003
1.759
3.32
A
A
.1875+.0000 -.0050
.0625
.125+.005 -.000
SECTION A-A
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±1/16
ANGULAR ±.5°
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005
INTERPRET GEOMETRIC TOLERANCING PER: ASME Y14.5-2018
MATERIAL
4140 Steel
QUANTITY
1
DXF
DO NOT SCALE DRAWING
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Daniel Choate
03/14/23
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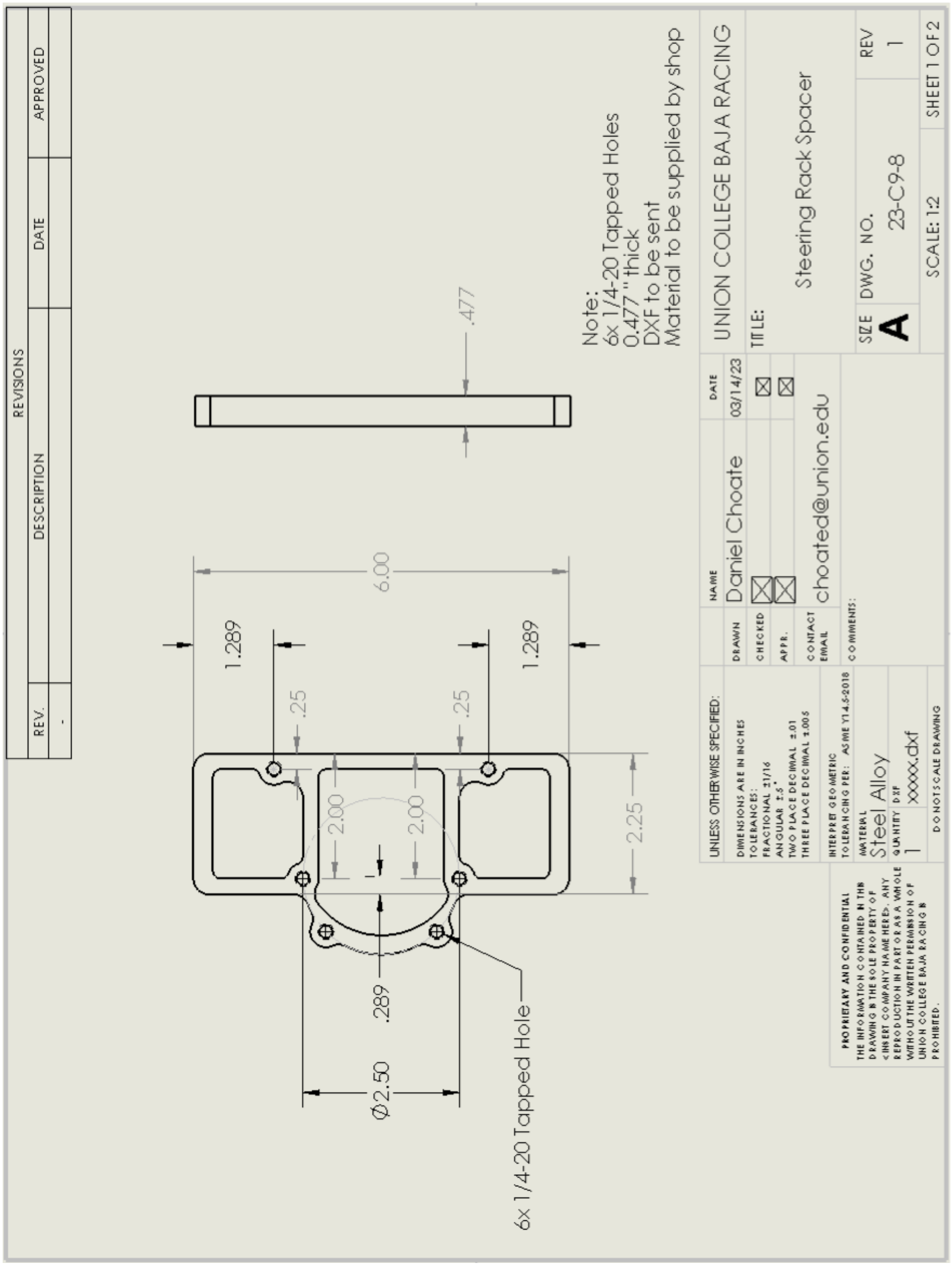
APPR.
CONTACT EMAIL
choated@union.edu
COMMENTS:
UNION COLLEGE BAJA RACING
TITLE:
Steering Rack Shaft with Key Cut
SIZE
A
DWG. NO.
23-C9-6
REV
2
SCALE: 1:1
SHEET 1 OF 2


REVISIONS
REV.
DESCRIPTION
DATE
APPROVED
-
1.289
.25
.25
1.289
6.00
2.00
2.00
2.25
Ø2.50
.289
6x 1/4-20 Tapped Hole
.477
Note:
6x 1/4-20 Tapped Holes
0.477" thick
DXF to be sent
Material to be supplied by shop
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±1/16
ANGULAR ±.5°
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005
INTERPRET GEOMETRIC
TOLERANCING PER: ASME Y14.5-2018
MATERIAL
Steel Alloy
QUANTITY
1
DXF
xxxx.dxf
DO NOT SCALE DRAWING
NAME
DATE
DRAWN
Daniel Choate
03/14/23
CHECKED
APPR.
CONTACT EMAIL
choated@union.edu
COMMENTS:
UNION COLLEGE BAJA RACING
TITLE:
Steering Rack Spacer
SIZE
A
DWG. NO.
23-C9-8
REV
1
SCALE: 1:2
SHEET 1 OF 2
PROPRIETARY AND CONFIDENTIAL
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Appendix B: Critical Calculations

Lewis Equation Calculations								
	inches	metric						
pressure angle	20	20						
teeth	16	16						
face width	0.75	19.05	in to m					
pitch diameter	1	0.0254	in to m					
module	0.0625	1.5875	CHECK (mm?)					
y	0.2761	0.2761						
velocity	3.14159265	0.03989823	rad/s to m/s					
Kv		0.99350181						
outer radius (gear)	1.125	0.028575	in to m					
R(sw)	5.5	0.1397	in to m		bearing force	torque over area	sy	bearing stress
Force applied	60	266.893297	lbf to N		660	4224	44000	8448
Torque	330	37.2849936	lbf-in to N-m				shear stress	bearing stress
Transmitted load (Wt)	293.333333	1304.81167	lbf to N			FOS	5.20833333	5.20833333
Bending Stress	22664.896	157291064	22813.1401	pa to psi				
yield strength			120000	psi				
Factor of Safety			5.260126379					

Appendix C: Cost Breakdown

Parts Order List - Daniel Choate				
Steering Rack				
Part Name	Part Number	Quantity	Price	Total
Metal Round Rack	McMaster - 2932N1	1	$ 67.42	$ 67.42
Metal Gear - 16 teeth	McMaster - 5172T63	1	$ 38.92	$ 38.92
High load ball bearing - 0.75 ID	McMaster - 2780T65	1	$ 34.17	$ 34.17
Low Friction Sleeve Bearing	McMaster - 1677K385	2	$ 10.08	$ 20.16
Press-fit Drill Bushing	McMaster - 8492A411	1	$ 17.42	$ 17.42
Steel Socket Head Screws, (50)	McMaster - 91251A542	1	$ 11.97	$ 11.97
Inline Ball Joint Linkage	McMaster - 8412K43	2	$ 13.95	$ 27.90
Aluminum Block	Albany Metal	1	$ 60.00	$ 60.00
TOTAL				$ 277.96

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